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# A review of the performance of double pass solar air heater

Sunil Chamoli[a,*], Ranchan Chauhan[a], N.S. Thakur[a], J.S. Saini[b]

[a] Centre For Excellence in Energy and Environment, National Institute of Technology, Hamirpur 177005, India
[b] Department of Mechanical and Industrial Engineering Indian Institute of Technology, Roorkee 247667, India

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ABSTRACT

Improvement of the thermal performance of a solar air heater can be obtained by enhancing the rate of heat transfer. The thermal efficiency of double pass solar air heater is higher in comparison to single pass with the concept involved of doubling the heat transfer area without increase in the system cost. Numbers of studies have been carried out on the performance analysis of double pass solar air heater provided with heat transfer augmentation techniques viz. using extended surfaces, packed bed, corrugated absorber were reported in the literature and found more increase in the thermal efficiency in comparison to conventional double duct solar air heater. These studies includes the design of double pass solar air heater, heat transfer enhancement, flow phenomenon and pressure drop in duct. This paper presents an extensive study of the research carried out on double pass solar air heater. Based on the literature review, it is concluded that most of the studies carried out on double pass solar air heater integrated with porous media and extended surfaces. Few studies were carried out with corrugated absorber. Further no study has been reported so far on double pass solar air heater with absorber plate artificially roughened from both the sides. Mathematical models based on energy analysis of some configurations of solar air heater have been discussed.



**Contents**



* Corresponding author. Tel.: +91 9816397033.
*E-mail address:* doon_84@yahoo.com (S. Chamoli).

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## 1. Introduction

Energy is available in multifarious forms and plays a significant role in world wide economic growth and industrialization. The growth of world population accompanied with rising material needs intensified the rate of energy usage. Continuous increase in energy usage characteristics of the past 50–100 years cannot continue indefinitely as demarcated energy resources of earth are exploitable.

On the other hand, environment degradation with use of fossil fuels is a menace to life in this earth. In view of world's depleting fossil fuel reserves and environmental threats, development of renewable energy sources received importance. Of many alternatives, solar energy stands out a conspicuous energy source for meeting the demand. It is considered as adamantine renewable energy source due to its huge potential. The freely available solar radiation provides an infinite and non polluting reservoir of fuel. The easiest way to utilize solar energy for heating applications is to convert it into thermal energy by using solar collectors. Solar water heaters and solar air heaters are flat plate collectors which used mostly for heating of water and air respectively. Solar air heater is compact and less complicated in comparison to solar water heater. Solar air heater is easy to fabricate with cheaper materials and is easy to use then solar water heater. Solar air heater produce hot air for any industrial or farmer level drying applications by using freely available solar energy, without using any conventional fuels like electricity, diesel, LPG, firewood, coal, etc., But it could be coupled with an existing conventional drying systems like Tray driers, tunnel driers, conveyor drier, FBD drier and bin drier operated by conventional fuels to save fuel consumption. The thermal efficiency of solar air heater is less due to low heat transfer capability between absorber plate and fluid flowing in the duct. To make solar air heater more efficient solar energy utilization system, thermal efficiency needs to be improved by enhancing heat transfer rate. It is reported in literature that the heat transfer rate can be exaggerated by the following: (a) use of a secondary heat transfer surface, (b) disruption of the unenhanced fluid velocity, (c) disruption of the laminar sublayer in the turbulent boundary layer. Under the method (c) i.e. disruption of laminar sublayer by using wires or ribs in forms of artificial roughness provided underside of absorber plate, method (a) includes the contact fluid surface area increase by using extended surfaces i.e. fins and by using packed bed i.e. a volume of porous media obtained by packing particles of selected material into a duct. In the packed bed solar energy penetrates to greater depths and is absorbed gradually depending on the density of packing. Many researchers carried out experimental investigation of heat transfer augmentation techniques i.e. extended surfaces, packed bed and corrugated absorber in double pass solar air heater but so far no work has been reported with artificially roughened double pass solar air heater duct. In the present article an attempt has been made in order to make a brief outline of the performance double pass solar air heater and the various methods those exaggerate their performance and mathematical models of some configurations of double pass solar air heaters are discussed.

## 2. Performance analysis of double pass solar air heater

It is required to analyze thermal performance of solar air heater for making an efficient design of a system. Thermal performance concerns with heat transfer process within collector. The different arrangement of the double pass solar air is shown in Fig. 1(a)–(c). Fig. 1(a) is considered for the thermal analysis and the design of such conventional system are described by Satcunanathan and Deonarine [1].

**Nomenclature**

| | |
|---|---|
| $A$ | area element |
| $B$ | height of solar air collector |
| $C$ | specific heat |
| $D$ | depth of air channel |
| $D_h$ | hydraulic diameter of air channel |
| $h$ | heat transfer coefficient |
| $k$ | thermal conductivity |
| $L$ | length of air collector |
| $Q$ | Heat collected |
| $m$ | mass flow rate |
| Nu | Nusselt number |
| Pr | Prandlt number |
| $R$ | recycle ratio |
| $r$ | fraction of mass flow rate |
| Re | Reynolds number |
| $S$ | incident solar radiation |
| $T$ | temperature |
| $t$ | time |
| $U$ | loss coefficient |
| $V$ | wind speed |
| $W$ | width of absorber |
| $x,y$ | space coordinates |
| $H$ | fin height |
| $z$ | distance from absorber plate |
| $N$ | number of fins |

*Greek letters*

| | |
|---|---|
| $\alpha$ | solar absorbtance |
| $\varepsilon$ | emissivity |
| $\delta$ | thickness |
| $\rho$ | reflectance |
| $\tau$ | transmitivity |
| $\eta$ | efficiency |
| $\Delta$ | ratio of channel thickness |

*Subscript*

| | |
|---|---|
| $a$ | ambient |
| $b$ | base plate |
| $c$ | convection |
| $f$ | fluid(air) |
| $g$ | glass cover |
| $i$ | inlet |
| $o$ | outlet |
| $p$ | absorber plate |
| $r$ | radiation |
| $l$ | lower channel |
| $u$ | upper channel |

### 2.1. Thermal performance

The double pass solar air heater are of mainly two types depending on the fluid flow direction namely counter or return flow double pass solar air heater and parallel pass double duct solar air heater.

The analysis of parallel pass arrangement i.e. Fig. 1(c) heat collected in the lower and upper channel per unit area is defined as

$$q_1 = h_{c,p-l}(T_p - T_l) + h_{c,l-b}(T_b - T_l) \quad (1)$$

$$q_2 = h_{c,g-u}(T_p - T_u) + h_{c,u-p}(T_p - T_u) \quad (2)$$

where heat collected by lower flow and upper flow are $Q_l$ and $Q_u$, respectively defined as

$$Q_l = m_l c_p (T_{l,fo} - T_{l,fi}) \quad (3)$$

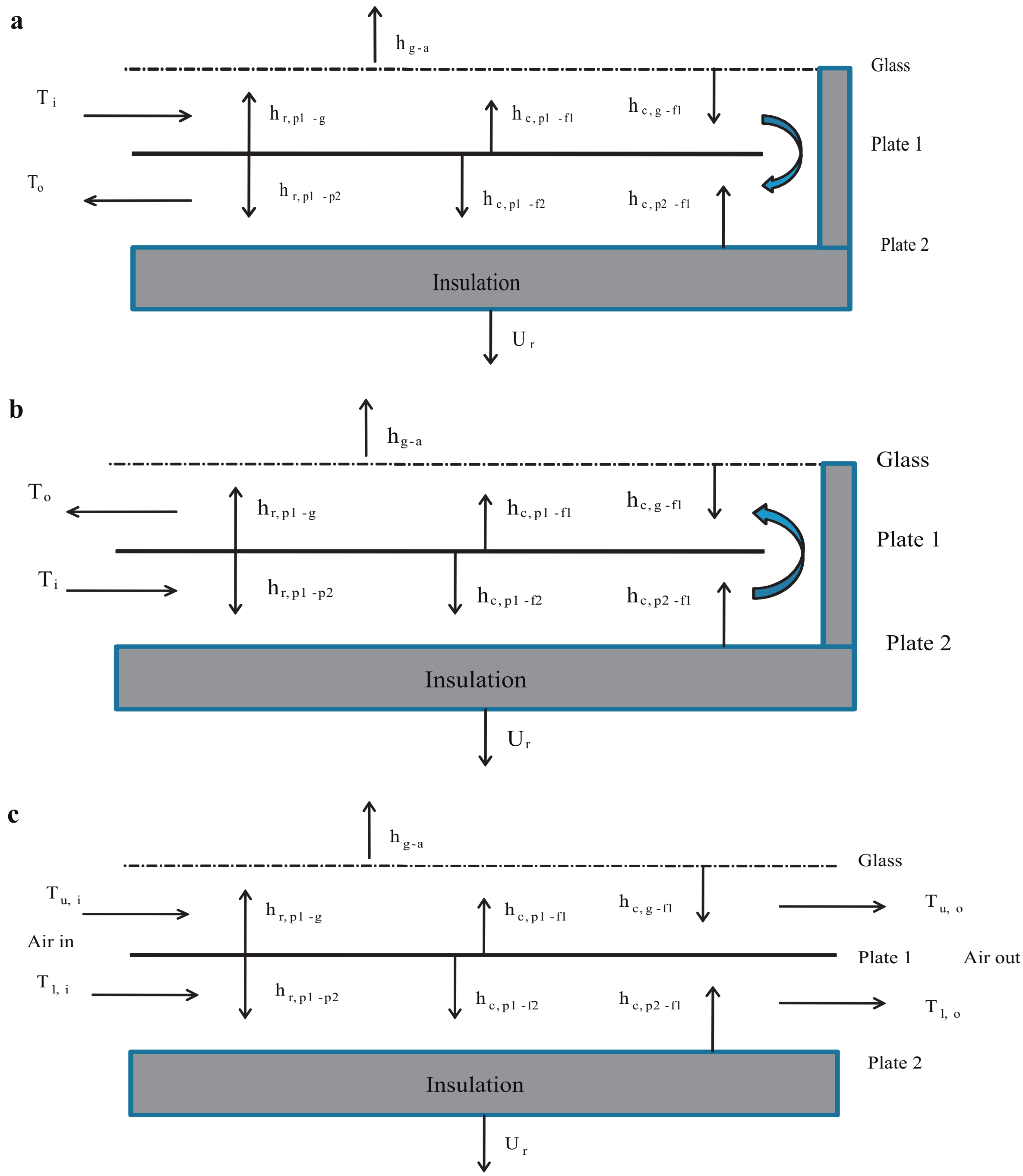



**Fig. 1.** (a) Schematic diagram of counter flow double pass solar air heater. (b) Schematic diagram of counter flow double pass solar air heater. (c) Schematic diagram of parallel pass double duct solar air heater [1].

$$Q_u = m_u c_p (T_{u,fo} - T_{u,fi}) \tag{4}$$

Total heat transfer $Q_t = Q_l + Q_u$ and efficiency is defined as

$$\eta = \frac{Q_t}{AS} \tag{5}$$

## 3. Mathematical models

A mathematical model of the closed solar air heaters is used in particular, to assist in interpreting the observed phenomena in the solar air heaters, to design the system, to predict the trends, and to assist in optimization. A review of the mathematical models for predicting solar air heater systems carried out by Tchinda [47]. The models have evolved to a point where several features of the process can be predicted, more effort is required before the models can be applied to define actual operating conditions as well as to further investigate new solar air heater. A review has been done on the theoretical modeling of some different designs of solar air heaters. Table 1 shows the heat transfer coefficients used in the following configured models.

### 3.1. Parallel pass solar air heater

The design of the parallel pass solar air heater is to optimize the heat transfer transportation in the solar air heaters. The design consists of a glass cover, absorber plate and bottom insulated plate.

**Table 1**
Heat transfer coefficients used in different mathematical models.

| Reference | Solar air heater system | Heat transfer coefficients |
|---|---|---|
| Forson et al. [15] | Parallel pass solar air heater | Radiative heat transfer coefficients from the absorber to the glazing, $h_{r,pc}$ and from the absorber to the bottom plate, $h_{r,pb}$ are given as $h_{r,pi} = \sigma(T_p^2 + T_i^2)(T_p + T_i)/(\varepsilon_p^{-1} + \varepsilon_p^{-1} - 1)$ where $i$ represents $c$, $b$. The heat transfer by radiation from the glazing to sky is given by $h_{r,cs} = \sigma\varepsilon_c(T_c^2 + T_s^2)(T_c + T_s)$. The forced convection heat transfer coefficient between parallel plates is given as: for laminar flow (Re < 2300) [36] $(hD_h/k) = 5.4 + 0.00190(\mathrm{RePr}D_h/L)^{1.71}/1 + 0.00563(\mathrm{RePr}D_h/L)^{1.71}$, for transition flow region (2300 < Re < 6000) [37]: $(hD_h/k) = 0.116(\mathrm{Re}^{2/3} - 125)\mathrm{Pr}^{1/3}[1 + (D_h/L)^{2/3}](\mu/\mu_w)^{0.14}$ and for turbulent flow (Re > 6000) [38] $(hD_h/k) = 0.018\mathrm{Re}^{0.08}\mathrm{Pr}^{0.4}$, $h_{rs} = \sigma\varepsilon_g(T_g^4 - T_s^4)/(T_g - T_s)$. Free convective heat transfer coefficient is given by: $h = k/D_h\{0.0158Re_{D_h}^{0.8} + (0.0181Re_{D_h} + 2.92)\exp(-0.0379x/D_h)$ where $Re_{D_h} = 2M/(1 + D/W)$. $U_b = [1 + 2(\delta_{\mathrm{in}} + D + \delta_g/W)][\delta_{\mathrm{in}}/k_{\mathrm{in}} + 1/h_w]^{-1}$. $U_{ej} = (UA)_{edge}/A_c = \{(k/x)S_j\}/\mathrm{W}$. |
| Ho et al. [8] | Double pass flat plate solar air heater with recycle | $h_1 = h'_1$, $h_2 = h'_2$. The forced convection heat transfer between two plates see Kays [39], for short conduit $h_1$ and $h_2$ see [40]: $\mathrm{Nu}_f = h_jD_e/k = 0.0158Re_f^{0.8}[1 + (D_e/L)^{0.7}](j = 1, 2)$, for the laminar flow see Heaton et al. [36], $1/h_b = 1/h_{B1s} + 1/h'_2 + l_B/k_B$ $1/U_{gls} = 1/h_{rg2s} + h_w + 1/h_{cg1g2} + h_{rg1g2}$ where $h_{cg1g2} = 1.25(T_{g1m} - T_{g2m})^{0.25}$ see [41], $h_{rpg1} = 4\sigma T_{bm}^3/[(1/\varepsilon_p) + (1/\varepsilon_R) - 1]$, $h_{rpr} = 4\sigma T_{am}^3/[(1/\varepsilon_p) + (1/\varepsilon_R) - 1]$, those between two glass covers see [42] and from cover 2 to ambient, $h_{rg2s} = \sigma\varepsilon_g(T_{g2}^2 + T_{sky}^2)(T_{g2} + T_{sky})$. |
| Naphon [18] | Double pass flat plate solar air heater with porous media | The radiative heat transfer coefficients see [15]. The convective heat transfer coefficient for air flowing over the outside surface of the glass cover is proposed by [40] $h_w = 5.7 + 3.8\,\mathrm{V}$, convective heat transfer coefficient for flowing in the channel is given by [43]: $\mathrm{Nu} = (hD_h/k) = 0.333\mathrm{Re}^{0.8}\mathrm{Pr}^{1/3}$ with $D_h = 4WB/2W + 2B$. |
| Naphon [28] | Double pass flat plate solar air heater with longitudinal fins | The convective heat transfer coefficient between channel see [28], for radiative heat transfer coefficient see [15]. |
| Ho et al. [7] | Multi-pass flat plate solar air heaters with external recycle | $h_1 = h'_1$, $h_2 = h'_2$, $h_3 = h'_3$ and $h_4 = h'_4$. Other coefficients see Ho et al. [8]. |
| Ramadan et al. [20] | Double pass solar air heater with porous media packed | $h_{cg1f1}$ $\mathrm{Nu}_{p1kf1}/D_e = 0.2Re_{p1}^{0.8}Pr^{1/3}$ $k_{f1}/D_e$ see [44], with $\mathrm{Re}_{p1} = m_fD_e/\mu A_{p1}$, $D_e = 2/3[\varepsilon D_{p1}/(1 - \varepsilon)]$, $\varepsilon$ is the porosity of packed bed ($\varepsilon < 1$): $\varepsilon = (V_{\mathrm{chan}} - V_{p1})/V_{\mathrm{chan}}$, $V_{\mathrm{chan}}$ and $V_{p1}$ are the total volume of the channel and the volume of the packed bed material. $D_{p1} = 6V_s^{1/3}n\pi$ see [45], $V_s$ = total volume of $n$ particles selected randomly. $A_{p1} = 6(1 - \varepsilon)/D_{p1}$ see [45], $h_{pf1} = h_{cg1f1}$ see [15], $h_{cp1f1} = [1/h_{p1f} + D_e/D_{p1}S_{p1}]^{-1}$ see [46] with $N_{p1f} = h_{p1f}D_e/k_{f1} = (0.255/\varepsilon)$ $Re_{p1}^{2/3}Pr^{1/3}$, $S_{p1}$ is the constant depends on the packed bed shape. |
| EI-Sebaii et al. [19] | Double pass solar air heater with porous media packed | The heat transfer coefficients see [20]. |

There are two air flow channels which one is located between the glass cover–absorber and another is located between the absorber and the bottom insulated plate as shown in Fig. 1(c). Forson et al. [15], obtained the energy transport equations for the glass cover, channel 1, absorber plate, channel 2, and the base plate is following:

At the cover:

$$M_gC_g\frac{\partial T_g}{\partial t} = \alpha_g S + h_{rpg}(T_p - T_g) + h_{cf-1g}(T_{f1} - T_g) - h_{cgw}(T_g - T_w) - h_{rga}(T_g - T_a) \tag{6}$$

For air flowing between cover and plate absorber:

$$M_{f1}C_f\frac{\partial T_{f1}}{\partial t} = \frac{-G_1C_f}{W}\frac{\partial T_{f1}}{\partial x} + h_{cpf1}(T_p - T_{f1}) - h_{cf1g}(T_{f1} - T_g) \tag{7}$$

For plate 1:

$$M_1C_p\frac{\partial T_p}{\partial t} = \alpha_p\tau_g S - k_p\delta_p\frac{\partial^2 T_p}{\partial x^2} - h_{rpg}(T_p - T_g) - h_{cpf2}(T_p - T_{f2}) - h_{rpb}(T_p - T_b) - h_{cpf1}(T_p - T_{f1}) \tag{8}$$

For air flowing between plate absorber and base plate:

$$M_{f2}C_f\frac{\partial T_{f2}}{\partial t} = \frac{-G_2C_f}{W}\frac{\partial T_{f2}}{\partial x} + h_{cpf2}(T_p - T_{f2}) - h_{cbf2}(T_b - T_{f2}) \tag{9}$$

For bottom plate:

$$M_bC_b\frac{\partial T_b}{\partial t} = -k_b\delta_b\frac{\partial^2 T_b}{\partial x^2} + h_{rpb}(T_p - T_b) - h_{cbf2}(T_b - T_{f2}) - h_b(T_b - T_r) \tag{10}$$

Along with following boundary and initial conditions

$$\left.\frac{\partial T_p}{\partial x}\right|_{x=0} = 0,\ \left.\frac{\partial T_p}{\partial x}\right|_{x=L} = 0,\ \left.\frac{\partial T_b}{\partial x}\right|_{x=L} = 0$$
$$T_f(x = 0) = T_{fi},\ T_{f2}(x = 0) = T_{fi}$$

### 3.2. Counter flow solar air heater

Ho et al. [8] developed the mathematical model for the counter flow solar air heater is shown in Fig. 2. In such collectors air is allowed to move faster than the simple collectors. The following assumptions were considered.

a) Temperatures of absorbing plate, bottom insulated plate and the fluid are only functions of air flow direction.
b) Glass cover and absorber plate do not absorb radiant energy.
c) The radiant energy absorb by the outlet cover is negligible.

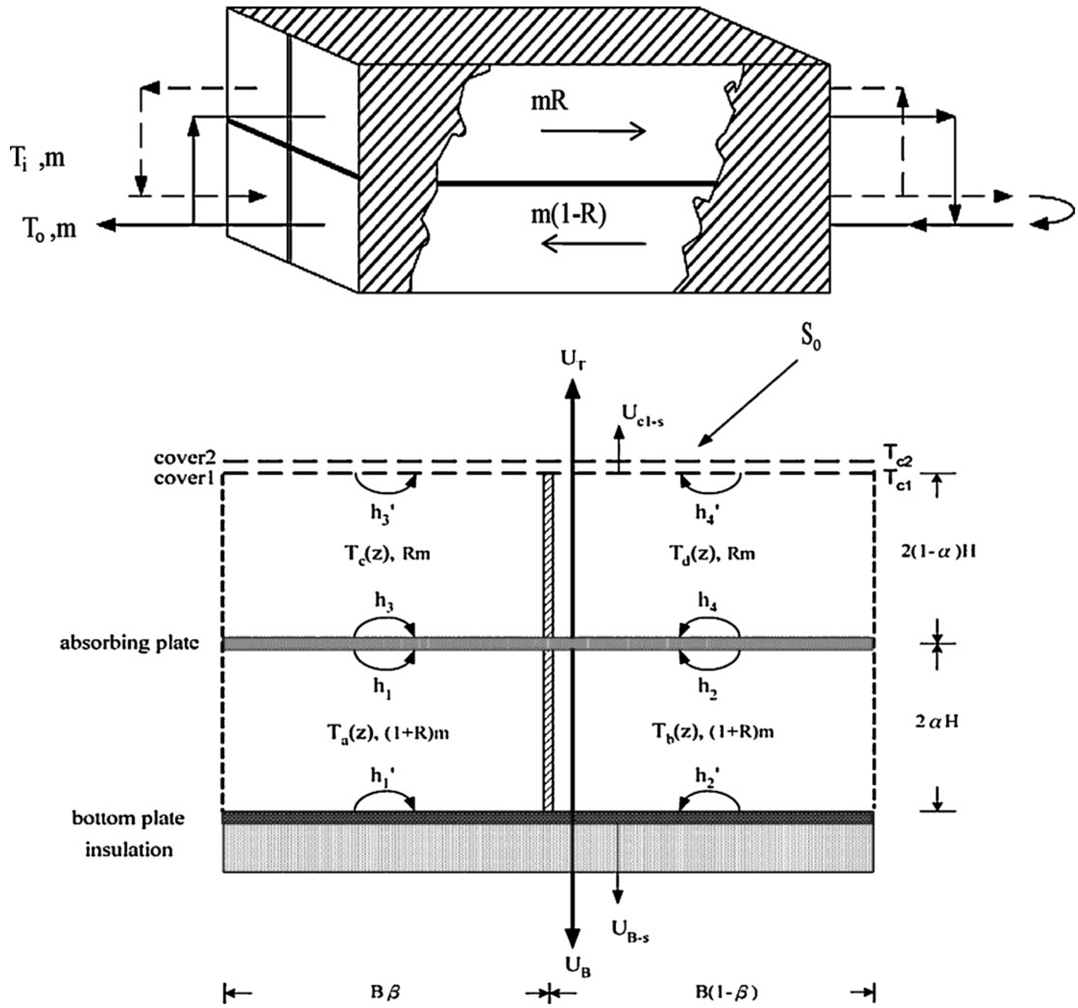



**Fig. 2.** Multipass flat plate solar air heater with recycle [7].

Considering above assumptions, energy equations of cover 1, absorber plate, bottom plate, channels are as following:

For cover 1:

$$h_{rpg2}(T_p - T_{g1}) + h'_1(T_b(z) - T_{g1}) = U_{g1s}(T_{g1} - T_s) \tag{11}$$

Absorbing plate:

$$\alpha_p \tau_{g1} \tau_{g2} S - h_T(T_p - T_s) + h_B(T_p - T_s) + h_1(T_p - T_b(x)) + h_2(T_p - T_a(x)) \tag{12}$$

Bottom plate:

$$h_{rpR}(T_p - T_R) + h'_2(T_a(x) - T_R) = U_{B1s}(T_p - T_s) \tag{13}$$

Bottom channel:

$$h_2(T_p - T_a(x)) - h'_2(T_a(x) - T_R) = \frac{(R+1)mC_p}{W}\frac{dT_a(x)}{dx} \tag{14}$$

Upper channel:

$$h_1(T_p - T_b(x)) - h'_2(T_b(x) - T_{g1}) = -\frac{(R+1)mC_p}{W}\frac{dT_b(x)}{dx} \tag{15}$$

In this the characteristic length is the equivalent diameter of the duct. For open conduit, lower channel, upper channel, they are respectively given by:

$$D_{e,o} = \frac{4HW}{2H+W}, \quad D_{ea} = \frac{4\Delta HW}{2\Delta H + W}$$

$$\text{and} \quad D_{e,b} = \frac{[4(1-\Delta)H]W}{[2(1-\Delta)H] + W} \tag{16}$$

and the air velocities of the open conduit, lower channel, upper channel used by Ho et al. [8] is given by:

$$v_o = \frac{m}{2HW\rho}, \quad v_a = \frac{(1+R)m}{2\Delta HW\rho}, \quad v_b = \frac{[m(1+R)]}{[2(1-\Delta)H]W\rho} \tag{17}$$

### 3.3. Double pass flat plate solar air heater with porous media

Naphon [18] studied theoretically the heat transfer characteristics and performance of double pass flat solar air heater with and without porous media. The mathematical model proposed in this study of design of the double pass single solar air collector with porous media is based on the model proposed by Naphon and Kongrtagool [6] and Wijeysundera [2]. The energy equations of the system are as follows:

At cover glazing:

$$\alpha_g S = h_{rgs}(T_g - T_s) + h_{cg-f1}(T_g - T_{f1}) + h_w(T_g - T_a) + h_{rgp}(T_g - T_p) \tag{18}$$

At absorber plate:

$$\alpha_p \tau_g S + h_{rgp}(T_g - T_p) + h_{cf1p}(T_{f1} - T_p) = h_{cpf2}(T_p - T_{f2}) + h_{rpb}(T_p - T_b) \tag{19}$$

At rear plate:

$$h_{cf2b}(T_{f2}-T_b)+h_{rpb}(T_p-T_b)=U_b(T_b-T_a) \quad (20)$$

At a fluid between absorber plate and a rear plate:

$$mC_p\frac{dT_{f2}}{dx}=K_p\frac{d^2T_{f2}}{dx}+h_{cf2p}(T_p-T_{f2})+h_{cf2b}(T_b-T_{f2}) \quad (21)$$

The implicit method of finite difference scheme had been employed to solve these models.

### 3.4. Double pass flat plate solar air heater with longitudinal fins

Naphon [28] has investigated heat transfer characteristics and entropy generation of the double pass flat plate solar air heater with longitudinal fins as shown in Fig. 4. The mathematical model developed with the energy balance equations as follows:

For glass cover:

$$\alpha_g S=h_w(T_g-T_a)+h_{cf1cg}(T_g-T_{f1})+h_{rgp}(T_g-T_p)+h_{rag}(T_g-T_a) \quad (22)$$

For the first air pass:

$$mC_f\frac{dT_{f1}}{dx}=h_{cf1g}(T_p-T_{f1})+h_{cf1p}(T_p-T_{f1})+\frac{N}{A_{frontal}}\int_{Z=0}^{Z=H}2Lh_1(T_{v1}-T_{f1})\,dz \quad (23)$$

For upper fin:

$$\frac{N}{A_{frontal}}\left(-kA\frac{dT_{v1}}{dz}\right)_{Z=0}=\frac{N}{A_{frontal}}\int_{Z=0}^{Z=H}2Lh_1(T_{v1}-T_{f1})\,dz \quad (24)$$

For absorber plate:

$$\alpha_g\tau_g S=h_{cf1p}(T_p-T_{f1})+h_{cf2p}(T_p-T_{f2})+h_{rgp}(T_p-T_g)+h_{rpb}(T_p-T_b)+\frac{N}{A_{frontal}}\left(-kA\frac{dT_{v1}}{dz}\right)_{Z=0}+\left(-kA_{sf}\frac{dT_{v2}}{dz}\right) \quad (25)$$

For lower fin:

$$\frac{N}{A_{frontal}}\left(-kA\frac{dT_{v2}}{dz}\right)_{Z=0}=\frac{N}{A_{frontal}}\int_{Z=0}^{Z=H}2Lh_2(T_{v2}-T_{f2})\,dz \quad (26)$$

For second air pass:

$$mC_f\frac{dT_{f2}}{dx}=h_{cf2g}(T_p-T_{f2})+h_{cf2p}(T_p-T_{f2})+\frac{N}{A_{frontal}}\int_{Z=0}^{Z=H}2Lh_2(T_{v2}-T_{f2})\,dz \quad (27)$$

For bottom plate:

$$h_{cf2b}(T_b-T_{f2})+h_{rpb}(T_p-T_b)+U_a(T_b-T_a)=0 \quad (28)$$

### 3.5. Multi pass flat plate solar air heaters with external recycle

Ho et al. [7] have developed a theoretical formulation for a multi pass solar air heater as shown in Fig. 2 with external recycle and investigate the recycle effect on efficiency. In this study Ho el al. have considered a recycled four pass solar air heater with vertical thickness $2\alpha H$ and $2(1-\alpha)H$ ($\alpha$ is the channel thickness ratio in vertical direction) and horizontal width $2\beta H$ and $2(1-\beta)H$ ($\beta$ is the channel thickness ratio in horizontal direction). The following assumptions were considered for developing mathematical model: (a) the temperatures of the absorbing plate, bottom plate and bulk fluid were functions of the flow direction only, (b) both glass covers and fluid does not absorb radiant energy, (c) the glass cover 2 absorbtance is negligible, the energy equations are as follows:

Cover1:

$$h_{rpg1}(T_p-T_{g1})+h'_3(T_c(x)-T_{g1})+h'_4(T_d(x)-T_{g1})=U_{gls}(T_{g1}-T_s) \quad (29)$$

Absorbing plate:

$$\alpha_p\tau_{g1}\tau_{g2}S-h_T(T_p-T_s)-h_B(T_p-T_s)-h_1(T_p-T_a(x))-h_2(T_p-T_b(x))+h_3(T_c(x)-T_p)+h_4(T_d(x)-T_p)=0 \quad (30)$$

Bottom plate:

$$h_1(T_a(x)-T_R)+h'_2(T_b(x)-T_R)+h_{rpR}(T_p-T_R)=U_{Bs}(T_p-T_s) \quad (31)$$

Channel a (flowing under the absorber plate) : $h_1(T_p-T_a(x))$

$$-h'(T_a(x)-T_R)=\left[\frac{(R+1)mC_p}{W\beta}\right]\frac{dT_b(x)}{dx} \quad (32)$$

Channel b : $h_2(T_p-T_b(x))-h'(T_b(x)-T_R)$

$$=\left[-\frac{(R+1)mC_p}{W(1-\beta)}\right]\frac{dT_b(x)}{dx} \quad (33)$$

$$\text{Channel c}: h_3(T_p-T_c(x))-h'_3(T_c(x)-T_{g1})=\left[\frac{-RmC_p}{W\beta}\right]\frac{dT_c(x)}{dx} \quad (34)$$

$$\text{Channel d}: h_4(T_p-T_d(x))-h'_4(T_d(x)-T_{g1})=\left[\frac{RmC_p}{W(1-\beta)}\right]\frac{dT_d(x)}{dx} \quad (35)$$

In this study, equivalent diameters are given as:

$$D_{e,a}=\frac{4\alpha HW\beta}{2\alpha H+W\beta},\ D_{e,b}=\frac{4\alpha HW(1-\beta)}{2\alpha H+W(1-\beta)},\ D_{e,c}=\frac{4(1-\alpha)HW\beta}{2(1-\alpha)H+W\beta},\ D_{e,d}=\frac{4(1-\alpha)HW(1-\beta)}{2(1-\alpha)H+W(1-\beta)} \quad (36)$$

and average velocities are given as:

$$v_a=\frac{m_a}{2\alpha HW\beta\rho},\ v_b=\frac{m_b}{2\alpha HW(1-\beta)\rho},\ v_c=\frac{m_c}{2(1-\alpha)HW\beta\rho},\ v_d=\frac{m_d}{2(1-\alpha)HW(1-\beta)\rho} \quad (37)$$

The above equations were solved numerically using an iterative procedure.

### 3.6. Double pass solar air heater with packed bed

The schematic diagram of double pass solar air heater with packed bed in the upper channel is shown in Fig. 3 Ramadan et al. [20] investigated heat transfer in double pass solar air heater with porous media in upper channel. The mathematical model has been developed taking into account the following assumptions: (a) the air heater operates under steady state condition, (b) the heat capacities of the glass covers, absorber, and back plates and insulation are negligible, (c) the temperature of flowing air was varied only in the flow direction. By taking account of all the above assumptions the energy balance equation written as follows:

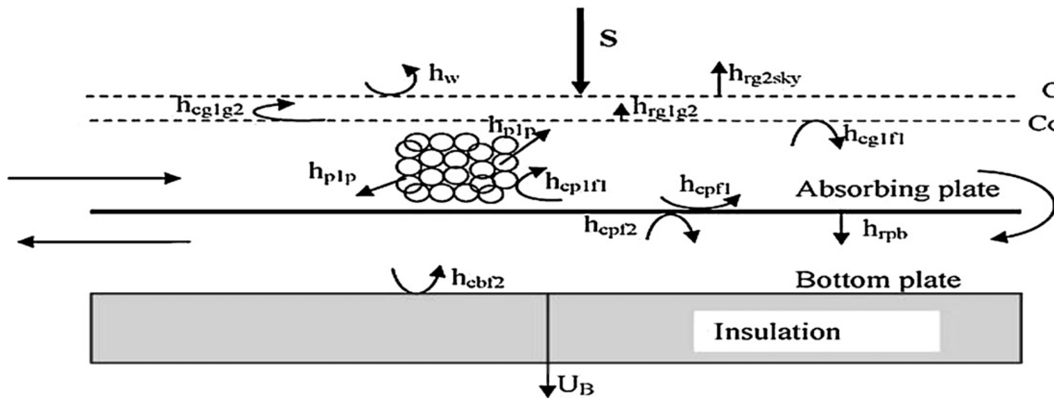



**Fig. 3.** Schematic view of energy flow of double pass with packed bed in upper channel [20].

Cover 2:

$$A_g\alpha_g S = A_g(h_{rg1g2}+h_{cg1g12})(T_{g2}-T_{g1}) + h_w A_g(T_{g2}-T_a) + A_g h_{rg2sky}(T_{g2}-T_{sky}) \tag{38}$$

Cover 1:

$$A_g\alpha_g\tau_g S = A_g(h_{rg1g2}+h_{cg1g12})(T_{g1}-T_{g2}) - h_{rp1g1}A_g(T_{p1}-T_{g1}) - h_{cg1f1}A_g(T_{f1}-T_{g1}) \tag{39}$$

Air flow in upper channel:

$$Wh_{pf1}(T_p-T_{f1}) + Wh_{cp1f1}(T_{p1}-T_{f1}) = m_{f1}C_f\frac{dT_{f1}}{dx} + Wh_{cg1f1}(T_{f1}-T_{g1}) + WU_s(T_{f1}-T_a) \tag{40}$$

Absorber plate:

$$A_p h_{rp1p}(T_{p1}-T_p) = A_p h_{pf1}(T_p-T_{f1}) + A_p h_{pf2}(T_p-T_{f2}) + A_p h_{rpb}(T_p-T_b) \tag{41}$$

Air flow in lower channel:

$$Wh_{pf2}(T_p-T_{f2}) + Wh_{f2b}(T_b-T_{f2}) = m_{f2}C_{f2}\frac{dT_{f2}}{dx} + WU_s(T_{f2}-T_a) \tag{42}$$

Back plate:

$$A_b h_{bf2}(T_{f2}-T_b) + A_b h_{rpb}(T_p-T_b) = A_b U_b(T_b-T_a) \tag{43}$$

The various heat transfer coefficient given in the above equations were calculated using correlation given by [42].

EI-Sebaii et al. [19] have proposed an investigation of heat transfer on a double pass solar air heater in which air is first flow through upper channel and then circulated in lower channel packed with porous media as shown in Fig. 4.The energy balance for the above system were as follows:

Cover 2:

$$A_g\alpha_g S = A_g(h_{rg1g2}+h_{cg1g12})(T_{g2}-T_{g1}) + h_w A_g(T_{g2}-T_a) + A_g h_{rg2sky}(T_{g2}-T_{sky}) \tag{44}$$

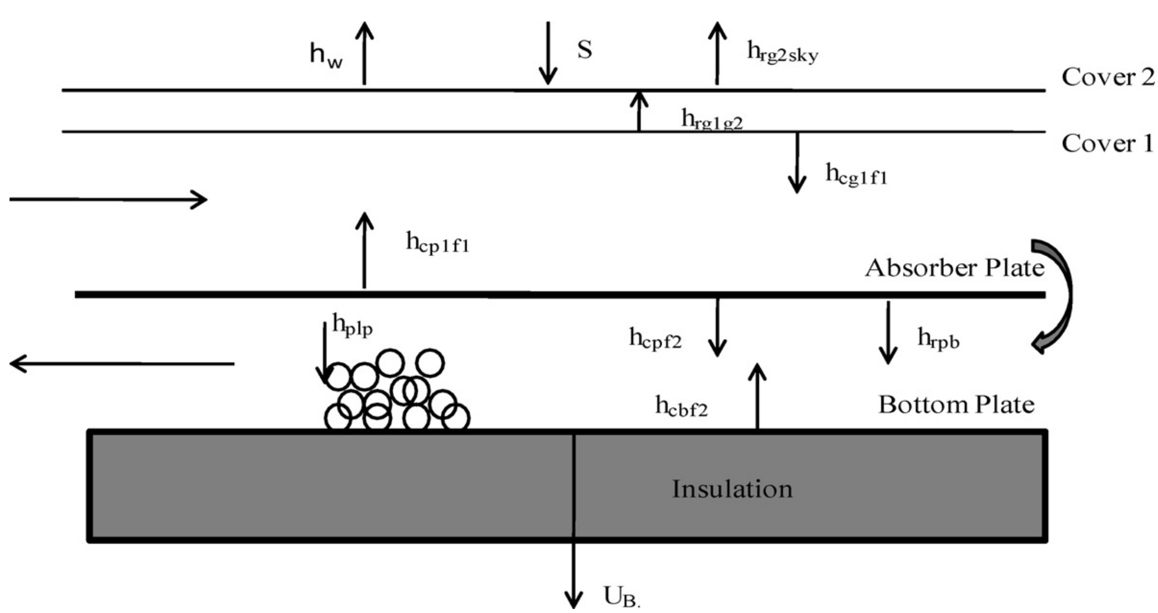



**Fig. 4.** Schematic view of counter flow solar air heater with porous media in lower channel [19].

Cover 1:

$$A_g\alpha_g\tau_g S - A_g(h_{rg1g2} + h_{cg1g2})(T_{g1} - T_{g2}) = -h_{rp1g1}A_g(T_{p1} - T_{g1}) - A_g h_{cg1f1}(T_{f1} - T_{g1}) \quad (45)$$

Air flow in upper channel:

$$Wh_{pf1}(T_p - T_{f1}) + Wh_{cp1f1}(T_{p1} - T_{f1}) = m_{f1}C_f\frac{dT_{f1}}{dx} + Wh_{cg1f1}(T_{f1} - T_{g1}) + WU_s(T_{f1} - T_a) \quad (46)$$

Absorber plate:

$$\alpha_p\tau_g{}^2A_pS = A_ph_{cpf1}(T_p - T_{f1}) + A_ph_{cpf2}(T_p - T_{f2}) + A_ph_{rpg1}(T_p - T_{g1}) + A_ph_{rpp1}(T_p - T_{p1}) \quad (47)$$

Air flow in lower channel:

$$Wh_{pf2}(T_p - T_{f2}) + Wh_{cp1f2}(T_{p1} - T_{f2}) = m_{f2}C_{f2}\frac{dT_{f2}}{dx} + WU_s(T_{f2} - T_a) + Wh_{bf2}(T_{f2} - T_b) \quad (48)$$

Back plate:

$$A_bh_{bf2}(T_{f2} - T_b) + A_bh_{rp1b}(T_{p1} - T_b) = A_bU_b(T_b - T_a) \quad (49)$$

The various heat transfer coefficients in the above equations were calculated using the correlations given by [44,45].

## 4. Smooth double pass solar air heater

### 4.1. Counter flow solar air heater

Satcunanathan and Deonarine [1] introduced the idea of double pass counter flow air heater. The air flows first between the space between glass cover and absorber plate and then induced through the duct. The thermal losses from one or more cover glass can suppress by using such systems. These systems are 10–15% more efficient then single pass system. A heat transfer model for two pass system is given by Wijeysundera et al. [2] and results were compared with single pass system. It is encountered that the open systems, with inlet fluid at ambient temperature, the double exposure systems are 10–15% more efficient then the single exposure for a wide range of operating conditions. It is articulated that the outer cover temperature is conterminous to ambient temperature compared to the system operated on single cover glass. A mathematical model for performance prediction of a two glass cover operated under single and double pass mode developed by Persad and Satcunanathan [3]. Study reveals that the performance of double pass is the independent of the length, over collector range considered and the critical plate spacing depends on temperature of the glass cover of collector. Chaudhary and Garg [4] modified the mathematical model proposed by Wijeysundera et al. [2] in order to study the effects of parameters like duct length, upper and lower air flow duct depths and mass flow rate of fluid in the system. It is found that the optimum duct depth, corresponding to minimal annual cost per unit solar energy gain, were different for different duct lengths and air flow rates. Recycling concept in air heater is introduced by Ho and Yeh [5] and reported that increasing the fluid velocity by using recycle in double pass parallel plate air heater enhance the heat transfer coefficient, resulting in improved performance. On increasing the fluid velocity, recycling produces an effect of remixing the inlet fluid with hot outgoing fluid. The mathematical model of Wijeysundera et al. [2] used by Naphon and Kongtragool [6] to field out numerically the effect of air flow rate on heat transfer characteristic and performance of the single and double pass solar air heater. The explicit technique of FDM was used to solve the models. Performance of multi pass solar air heater with external under countercurrent flow operations was studied by Ho et al. [7]. Study shows that the recycle effect increases the collector coefficient due to the increasing fluid velocity effect and may compensate for the temperature difference decrement. The recycle effect increases convective heat transfer rate and reduce the temperature driving force and thus the thermal performance increases and further improvement in collector efficiency is obtained when channel thickness ratio deviates from 0.5. The recycle effect on double pass flat solar air heater also studied by Ho et al. [8] is shown in Fig. 5 found an increase in convective heat transfer rate and reduces the temperature driving force and thus increases thermal performance. It explained the two effects originated with recycle; the first associated with the fluid mixing of hot and cold air which increases heat loss and the second is high rate of heat transfer thus the effect overcome the come and tends to enhance thermal performance of solar air heater, the performance decreases when the channel height deviates from 0.5 and 28–95% increase in collector efficiency improvement in comparison to the sub collectors in series.

A theoretical investigation of double pass solar air heater with constant heat flux by using an eigen function expansion in terms of power series for the homogeneous part and an asymptotic solution for the non homogeneous part was carried out by Ho et al. [9]. Study shows that the heat transfer efficiency increases with increasing graetz number and as sheet location is moved from 0.5. The mathematical model can be used in any geometry with more general boundary conditions of multi stream or multiphase problem and to predict the wall temperature distribution of the double pass systems. Ho et al. [10] experimentally and theoretically investigates a system where absorber plate divides the duct into two channels provided with external recycling. Experimental and theoretical results are represented graphically and compared with that in the downward-type single-pass solar air heaters of the same size without recycling. Considerable improvement in heat transfer was observed by employing double-pass operations with external recycling and fin attached over and under the absorbing plate. The influences of recycle ratio and absorbing plate location on

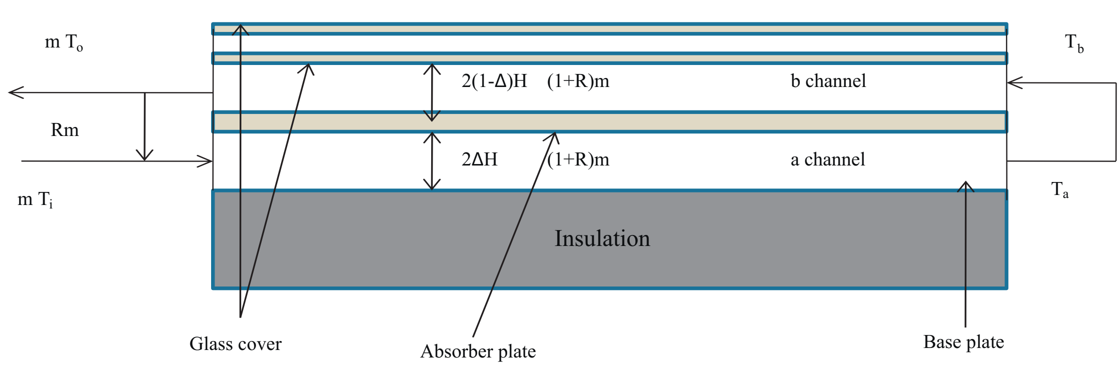



**Fig. 5.** The recycle effect on double pass flat solar air heater [8].

the heat-transfer efficiency enhancement as well as on the power consumption increment have been also delineated

### 4.2. Parallel pass double duct solar air heater

In parallel pass solar air heater the air flow is from upper and lower of the absorber plate simultaneously, in which there are reduced thermal losses in comparison to conventional system. The non porous double flow solar air heater system studied by Sodha et al. [11] and the study reveals that when air flows above and under the absorber plate, thermal performance is better comparative of single pass system. Mathematical model and solution procedure for predicting the thermal performance of a solar air heater for four different configurations: Type-1 single channel with single air blow between top glass and bottom absorber plate, Type-2 single channel with single air flow between absorber and base plate with no glass cover, Type-3 double channel with single air flow between absorber and bottom plate and Type-4 double channel with double air flows between glass and absorber plates was developed by Ong [12]. The model predicts the mean temperature of the heated air near the inlet of the collector and under predicts near the outlet. An experimental investigation in order to predict the thermal performance of a system with simultaneous air flows over and under the absorber plate was carried out by Yeh et al. [13], in order to study the fraction of mass flow rate in upper and lower channel. A considerable improvement in collector performance of double pass over single pass at same flow rate and in order to maximize the thermal performance of double pass collector the mass flow rates in both channel must be same. A model of single pass double duct solar air heater is given by Hegazy [14]. The study reveals that the channel depth to length ratio should be 0.0025 for both free and forced convection. The optimum value is considered and a theoretical study on collector shapes is conducted and the results shows for a constant area, variables width collectors exhibit performance behavior similar to constant width ones, but with a marginal decrease in collector efficiency. Forson et al. [15] Developed mathematical model for single pass double duct solar air heater (SPDDSAH) which will be a design tool capable of predicting the incident solar radiation, heat transfer coefficient, mean air flow rates, mean air temperature and relative humidity at the exit. It was found that on increasing the top to bottom channel ratio is a cost effective way of optimizing the thermal performance of SPDDSAH.

## 5. Double pass packed bed systems

### 5.1. Double pass solar air heater with packing material

There are two problems associated with the thermal performance of solar air heater. One is the heat transfer from the metallic plate to the flowing air is very less as low thermal conductivity of air. The second associated with the low heat transfer due to small thickness of absorber plate. This is increased by using a porous bed absorber. As the voids exits in the porous bed, solar radiation impinges to a greater depth and thus relatively absorbs there. The packing provides an increase in the turbulence which results a high rate of heat transfer. The packing material also acts as a heat storing media. A numerical study given by Mohamad [16] for investigation of the performance of double pass counter flow solar air heater with and without porous packed bed and its comparison with single and double glazing conventional solar air heaters. It was reported that thermal efficiency of a counter flow air heater without porous matrix is between 10 and 18% higher than the double glazing conventional air heater. The thermal efficiency of single glazing conventional air heater is between 18 and 25% lower than the counter flow air heater without porous matrix. An indoor test conducted by Sopian et al. [17] on the double pass solar collector with and without steel wool as porous media in the second channel. The collector had only single glass cover and a blackened metal absorber and the material used as the porous media was steel wool. Various combinations of upper and lower channel were considered for the investigation. The porous media in the second channel results an increase in outlet temperature and 20–60% increase in the thermal efficiency was reported. The heat transfer characteristics and performance of double pass solar air heater with and without porous media in the lower channel was studied by Naphon [18] as shown in the Fig. 6. The mathematical model developed in the study based on the systems of Wijeysundera et al. [2]. The effect of thermal conductivity of the porous media on the heat transfer characteristics was studied and the results validated with experimental results of Sopian et al. [17].

An experimental investigation of the thermal performance of double glass, double pass solar air heater with packed bed (DGDPSAHPB) was carried by EI-Sebaii et al. [19] to study the effect of mass flow rate and porosity on the air outlet temperature, thermal output power, pressure drop and thermo hydraulic efficiency. Study reveals that the heat transfer is enhanced by using gravel instead of lime stone as a packed bed above the absorber plate and the annual averages of the outlet temperature and thermo hydraulic efficiency of the DGDPSAHPB are about 16.5% and 28.5% higher than those of DGDPSAH. A theoretical and experimental investigation of double glass double pass solar heater with packed bed (DPSAHPB) above the heater absorber plate carried by Ramadan et al. [20]. The lime stone and gravels were used as packed bed materials and effect of mass flow rate of air and porosity of packed bed material was studied and it was found that for increasing the outlet temperature of the flowing air after sunset, it is advisable to use higher masses and low porosities. The thermo-hydraulic efficiency was found to increase with increasing mass flow rate until the flow rate of 0.05 kg/s beyond which the increase in efficiency becomes insignificant and operating range of mass flow rate is set to 0.05 kg/s or less in order to have low pressure drop across

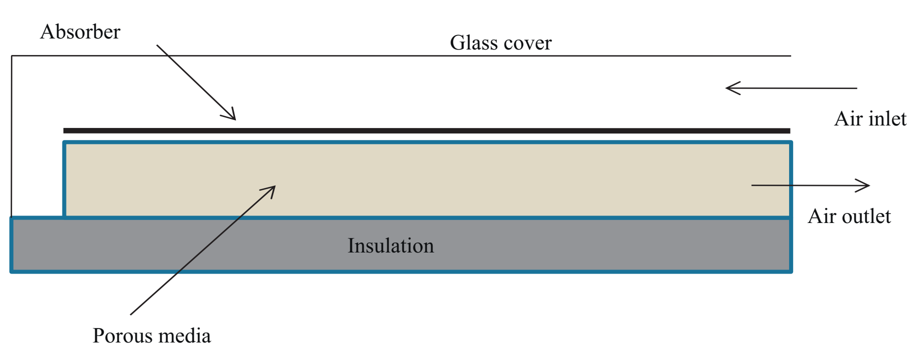



**Fig. 6.** Double pass solar air heater with porous media in the lower [18].

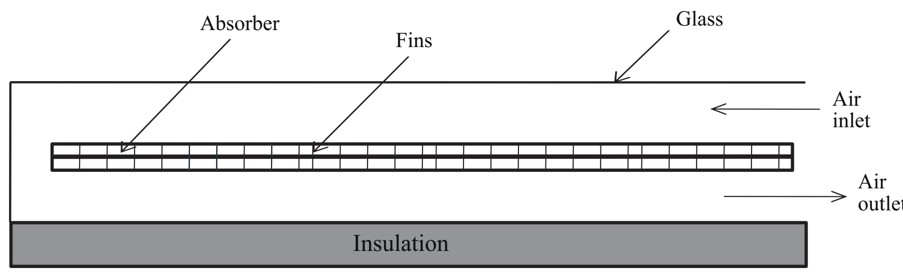



**Fig. 7.** Double pass solar air heater with longitudinal fins [28].

the system. Theoretical and experimental investigation of double pass solar heater with porous material packed in the lower channel carried by Sopian et al. [21]. The porous media arranged in different porosities to increase heat transfer, area density and the total heat transfer rate. Heat transfer and pressure drop relations have been developed for air flow through porous media. Porous media in the second channel increases the heat transfer coefficient and thus improved thermal efficiency. The thermal performances of single and double pass solar air heaters with steel wire mesh layers instead of a flat absorber plate were investigated experimentally by Aldabbagh et al. [22]. The results indicate that the efficiency increases with increasing the mass flow rate with in the range of 0.012–0.038 kg/s. For the same range of flow rate, efficiency of double pass is 34–45% higher than single pass and the maximum efficiency attained in single and double pass is 45.93% and 83.65%, respectively for the flow rate of 0.038 kg/s. Thermal performance of single and double pass solar air heater with fins attached and steel wire mesh as absorber was carried out experimentally by Omajaro and Aldabbagh [23]. Effect of air mass flow rate over the range of 0.012–0.038 kg/s on the outlet temperature and thermal efficiency was studied for the bed height of 7 cm and 3 cm of lower and upper channel, respectively. The efficiency of double pass was 7–19.4% more than of single pass system. The maximum efficiency attained at the flow rate of 0.038 kg/s and found 59.62% and 63.74% for single and double pass system, respectively. The proposed system of Mohamad [16], Aldabbagh et al. [22] is modified by Ramani et al. [24] and studies carried out on the system with air forced to flow in the lower channel packed with wire mesh matrix as porous media. The predicted values of the system validated with the outdoor studies and it was found that the efficiency of double pass with porous is 25% higher than without porous and 35% higher than the single pass system. An experimental study of double pass solar air heater with packing material on the upper channel was carried out by Dhiman et al. [25] and developed an analytical model was developed that describes the various temperatures and heat transfer characteristics of parallel flow packed bed solar air heater, employed to study the effects of mass flow rate and varying porosities of packed bed material on its thermal performance. PFPBSAH thermal efficiency was 10–20% higher than the conventional double pass system. The thermal performance of a double pass solar air heater with 2, 4, and 6 fins attached was investigated experimentally by El-khawajah et al. [26] and wire mesh layers were used between the fins instead of an absorber plate. The results indicate that the efficiency increases with increase in mass flow rate for the range of the flow rate used in this work between 0.0121 and 0.042 kg/s. Moreover, the maximum efficiency was obtained by using 6 fins at the same mass flow rate. The maximum efficiency obtained for the 2, 4, 6 fins of SAH were 75.0%, 82.1% and 85.9%, respectively for the mass flow rate of 0.042 kg/s. In addition, the maximum average temperature difference between the inlet and the outlet, $\Delta T$, for the SAH with 6 fins was the highest for the same mass flow rates compared to 2 and 4 fins SAH. The maximum average and instantaneous peak of $\Delta T$ obtained were 43.1 °C and 62.1 °C, respectively for the 6 fins SAH when the mass flow rate was 0.0121 kg/s.

## 6. Double pass system with extended surfaces

### 6.1. Counter flow with extended surfaces

Experimental investigation of heat transfer characteristics of a double flow channel with fins attached on the upper and lower sides of absorber plate was carried out by Yeh et al. [27] and fins of rectangular profile of height 5.5 cm and pitch of 6 cm were attached to the absorber plate. Both the theoretical and experimental investigation show that the optimal ratio of air flow rates in both the flow channels for maximum efficiency is unity i.e. both the channels have same mass flow rates. The double pass air heater with fins attached show a considerable increase in the performance as compared to the smooth one. A numerical study on the performance and entropy generation of the double pass solar air heater with longitudinal fins shown in Fig. 7 was carried by Naphon [28] and a mathematical model was developed for the prediction of heat transfer characteristics for the range of mass flow rate 0.02–0.1 kg/s. The effect of height and number of fins on entropy generation was considered and it was found that thermal efficiency increases with increasing the height and number of fins whereas the entropy generation was inversely proportional to the height and the number of fins.

Ho et al. [29] Carried out experimental and theoretical investigation of a double pass solar air heater with fins attached as shown in Fig. 8 and the experimental and theoretical results were compared

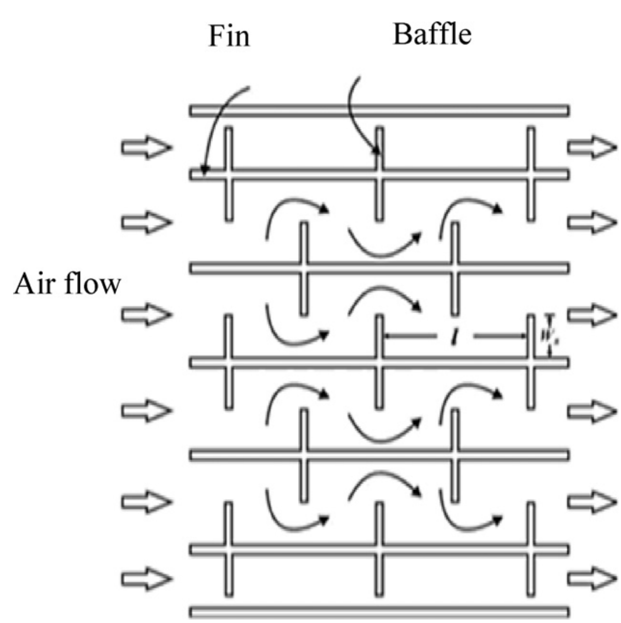



**Fig. 8.** Double pass solar air heater with fins and baffles [29].

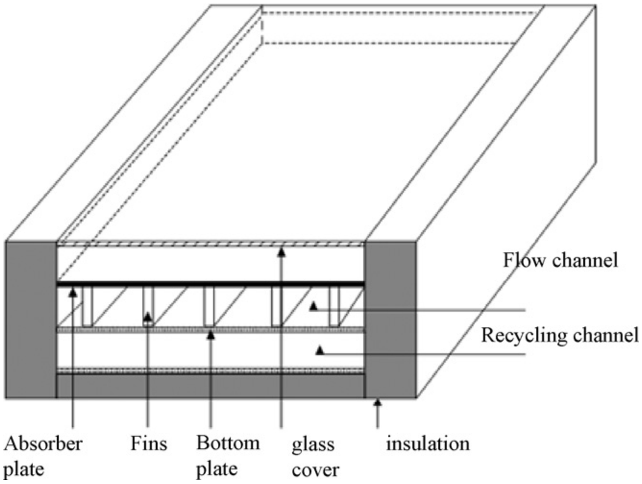



**Fig. 9.** Solar air heater with internal fins attached [30].

with downward type single pass solar air heaters of the same size without recycling. The study reveals that the baffled double pass solar air heater is a feasible design to improve collector efficiencies with recycles operations. As baffle create high turbulence and extending the heat transfer area and thus a high rate of heat transfer coefficient is achieved. It was noticed that on increasing the mass flow rate and recycle ratio is found to be beneficial for collector efficiency improvement; higher energy dissipation occurs at the same time when operating at high mass flow rate and recycling ratio.

A theoretical investigation has been carried out by Yeh and Ho [30] in order to study the influence of external recycle on the efficiency of solar air heater with internal fins attached to it is shown in Fig. 9. The external recycle produces two effects one is the increase in fluid velocity to decrease the heat transfer resistance while the second one is the lowering the driving force of heat transfer, due to remixing at the inlet, that a bad for performance. It was found that there will be considerable amount of efficiency increment if the operation is carried out with external recycle to suppress the undesirable effect. It was noted that enhancement increases with increase in reflux ratio, especially for operating at low flow rates with higher inlet temperatures. A theoretical investigation of the collector efficiency of upward type double pass solar air heater with fins attached and with external recycling was carried by Ho et al. [31] and a mathematical model was developed for predicting the heat transfer characteristics of the system. The equations for theoretical prediction of the useful heat gain, collector efficiency and outlet temperature were derived. It is found that desirable effect of increasing the fluid velocity using recycling to overcome the undesirable driving force decreases (temperature difference) for heat transfer due to remixing at the inlet, more than 80% improvement in collector efficiency is obtained using recycling operation.

### 6.2. Parallel pass flow with extended surfaces

Experimental energy and exergy analysis for a novel flat plate solar air heater with different obstacle in fluid flow was carried out by Esen [32], the operating parameters that are to be measured for double flow air heater were inlet and outlet temperature, the absorbing plate temperature, the ambient temperature and the solar radiation. It was found that double flow solar air heater increases the heat transfer area and thus the heat transfer rate, thermal and exergetic efficiencies.

Ozgen et al. [33] experimentally investigated absorber plate of double pass channel solar air heater. Three arrangements were used for absorber plate: (a) cans with staggered arrangement, (b) cans with inline arrangement and (c) plane duct without cans is shown in Fig. 10. The range of mass flow rate over which experiments performed was 0.03 and 0.05 kg/s. The double flow solar air heater with aluminum cans increases the heat transfer area and thus the convective heat transfer and the optimum value of collector efficiency was obtained for staggered arrangement at the mass flow rate of 0.05 kg/s.

Theoretical and experimental investigation of double pass finned plate solar air heater was carried out by EI-Sebaii et al. [34]. The numerical calculations were carried out for Tanta (Latitude 30° 47′N) prevailing weather conditions. The effect of mass flow rates of air on pressure drop, thermal and thermo hydraulic efficiencies of double pass finned plate and v corrugated absorber plate were studied. It was observed that the outlet temperature of DPVCPSAH is 2.1–9.7% higher than that of DPFIPSAH. DPVCPSAH was 9.3–11.9% efficient than that of DPFIPSAH and maximum thermo hydraulic efficiency of DPVCPSAH was 17.4% higher than that of DPFIPSAH and optimum values of thermo hydraulic efficiency for both the system are obtained at 0.0125 and 0.0225 kg/s. EI-Sebaii et al. [35] carried out experimental and theoretical investigation of double pass flat and v-corrugated plate solar air heater for Tanta (Latitude 30° 47′N) prevailing weather conditions. The effect of mass flow rate of air on pressure drop, thermal and thermo hydraulic efficiencies of double pass flat plate and v corrugated absorber plate was studied. Study reveals that the outlet temperature of DPVCPSAH is 5% higher than that of DPFPSAH and DPVCPSAH is 11–14% more efficient than that of DPFPSAH. The thermo hydraulic efficiency of DPVCPSAH was 14% higher than that of DPFPSAH and optimum values of thermo hydraulic efficiency for both the system are obtained at 0.02 kg/s.

## 7. Conclusion

Based on the review of the literature on double pass solar air heater system, it has been found that double pass solar air heater were widely investigated both analytically and experimentally. A number of studies have been carried out in order to investigate the effect of various parameters on the performance of double pass solar air heater. The mass flow rate of air and packing material




**Fig. 10.** Double pass channel solar air heater. Three arrangements used for absorber plate (I) cans with staggered arrangement (II) cans with inline arrangement (III) plane duct without cans [33].

porosity are considered the important parameters that affect the performance of the double pass solar air heater packed bed systems. The recycle ratio is found an important parameter that affect the performance of double pass solar air heater with external recycle and it observed that the maximum efficiency is obtained if the channel depths and mass flow rate of air is same in both the upper and lower duct. A number of analytical and experimental studies have been carried out with packed bed, fins integrated double pass solar air heater which shows significant increase of the performance compared to the conventional system. Few studies have been reported with corrugated absorber surface. Further, no study with artificially roughened double pass solar air heater is reported so far in order to compare the heat transfer characteristics and friction factor with conventional system. There is tremendous scope for future study of double pass solar air heater integrated with roughness element on both upper and lower side of the absorber surface. The information presented here will be beneficial for beginners in this area of research.

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